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PREDICTING CATASTROPHIC OUTSIDE DIAMETER INITIATED FATIGUE FAILURE OF THICK-WALLED CYLINDERS USING LOW CYCLE FATIGUE DATA

JOSEPH A. KAPP

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US ARMY ARMAMENT RESEARCH AND DEVELOPMENT CENTER
LARGE CALIBER WEAPON SYSTEMS LABORATORY
BENÉT WEAPONS LABORATORY
WATERVLIET N.Y. 12189

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A procedure is presented to predict fatigue failure of thick-walled cylinders containing discontinuities at the outside diameter (OD). Both crack initiation life and crack growth are considered. The elastic-plastic strains at an OD discontinuity are estimated using elastic analysis and stress concentration factors. The strain estimates are then used in conjunction with low cycle fatigue data to determine the initiation life. Crack growth life is estimated

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20. ABSTRACT (CONT'D)

by integration of a power law relationship. The results obtained by using this analysis method compared to measured fatigue life data for several OD initiated failures in thick-walled cylinders agrees to within about ten percent.

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INTRODUCTION

Thick-walled cylinders are often autofrettaged to enhance their fatigue lives (ref 1). The greater fatigue life results because of the compressive residual stresses at the inner diameter (ID). If the cylinder also contains a discontinuity at the outside diameter (OD) where the residual stresses are tensile, the combination of stress concentration and tensile operating and residual stresses often results in shortened fatigue life due to crack initiation at the OD.

Several attempts have been made to model this behavior experimentally (refs 2-4) and analytically (ref 5). This report presents a two-step prediction method which results in very accurate estimates of total fatigue life. The first step is to determine the crack initiation life through the use of low cycle fatigue data. A small semi-circular crack is then assumed to exist. Using known stress intensity factor solutions corrected for curvature, crack propagation life is predicted by integrating a power law relationship that accounts for mean stress effects (ref 6). The total fatigue life is then the sum of the initiation life and the propagation life. Fatigue lives predicted by this method agree very well with the measured lives of several thick-walled cylinders that fail from the OD.

PROCEDURE

Two different OD notch configurations shown in Figure 1 were studied, a single notch, Type A, and four notches, Type B. We assume that the material at the roots of these notches responds to service loading in the same manner

References are listed at the end of this report.

as a smooth cylindrical low cycle fatigue sample. By estimating the strain range that the material at the notch tip is subjected to, we should be able to determine the number of cycles to initiate a crack. This is accomplished with the assistance of the strain range versus cycles-to-failure plot shown in Figure 2. The plot was developed using smooth cylindrical specimens of the same steel used to manufacture the pressure vessels studied.

The three curves in Figure 2 are for three different strain ratios ($R = \varepsilon_{min}/\varepsilon_{max}$). This was included because the presence of autofrettage residual stresses will change R for the same internal pressure cycle. The three conditions tested were R = 0, no autofrettage; R = 0.5, high tensile residual stress; and R = 0.25, in between. The results show that the longest life was obtained with R = 0, the shortest with R = 0.25, and for R = 0.5, cycles-to-failure fell between these two maximina, although at large values of strain range there was no difference in the cycles-to-failure. The reason for these differences is explained by the stabilized cyclic strain curves that developed. When $\Delta\varepsilon$ was large there was essentially no difference in the hysteresis loops developed, thus no measurable difference in life, whereas the cyclic stress-strain behavior developed at smaller strain ranges was different. Often under these conditions, the cyclic stress-strain curves were simply elastic loadings and unloadings.

Figure 3 shows the actual results for $\Delta\varepsilon$ = 0.5 percent. The cumulative damage that results in failure of the specimen is a combination of two factors, the initial damage from the first loading to maximum strain and the cyclic loading. We would expect that the larger the initial strain the shorter the life, and the greater the amount of compression during cyclic loading the shorter the life. Because of the strain hardening of the steel, the R = 0 sample saw the most compression and the least initial strain, conversely, R = 0.5 had the least compression and the most initial strain, while the R = 0.25 saw intermediate values of each. The results of these tests suggest that the combination of two factors contributing to life is synergistic with the R = 0.25, the in between case being the worst.

Relating these low cycle fatigue results to OD failure was accomplished in the following manner. Two parameters had to be determined: strain range $\Delta\varepsilon$ and R. By using an elastic analysis these two values can be estimated. The minimum stress in an autofrettaged cylinder is $k_t \sigma_A$, where k_t is the stress concentration factor and σ_A is the elastic residual stress. Similarly, the maximum stress would be the minimum stress plus $k_t \sigma_p$, where σ_p is the stress due to pressure. The strain applied to the material at the notch tip then are estimated as

$$\varepsilon_{min} = \frac{\sigma_{min}}{E} = \frac{k_t \sigma_A}{E} \tag{1}$$

$$\varepsilon_{max} = \varepsilon_{min} + \frac{k_t \sigma_p}{E} \tag{2}$$

where E is Young's modulus for the material.

Using Eqs. (1) and (2) both the $\Delta\varepsilon$ and R parameters are easily calculated; $\Delta\varepsilon = \varepsilon_{max} - \varepsilon_{min}$, and $R = \varepsilon_{min}/\varepsilon_{max}$. For the cylinders considered below, this technique was used and k_t was either determined by finite element results available in the literature or by a Neubers Diagram (ref 7), and E was 207 GPa. Once $\Delta\varepsilon$ and R were calculated, the number of cycles to crack initiation was determined by interpolation of three curves in Figure 2.

Although crack initiation may consume a significant portion of the useful fatigue life of an OD notched thick-walled cylinder, the contribution of crack propagation must also be considered. To model crack growth, the following assumptions were made. At crack initiation a small semi-circular crack was produced. This crack was assumed to act as a crack with a depth equal to the notch depth and was semi-circular. The stress intensity factor for the crack was assumed to be the stress intensity factor for a straight-fronted crack of the same depth divided by the complete elliptical integral of the second kind to account for the curvature.

$$K_{(\text{curved crack})} = \frac{K_{(\text{straight crack})}}{\Phi} \quad (3)$$

The K solution for an OD cracked pressurized cylinder and for an OD cracked cylinder with autofrettage residual stress have been developed elsewhere (ref 5). The value of Φ used was that for a semi-circular crack which is 1.5708. Propagation of such a crack was assumed to follow a constitutive law given as (ref 6):

$$\frac{dc}{dN} = A\, K_{max}^{n} \quad (4)$$

where dc/dN is the crack growth rate, K_{max} is the maximum occurring stress intensity factor during a fatigue cycle, and A and n are material properties. Using Eq. (4) accounts for the mean K effect on crack propagation and has been shown to be the most conservative growth law for crack growth in OD cracked autofrettaged cylinders (ref 5). The crack growth life was determined by integration of Eq. (4).

RESULTS AND DISCUSSION

The results obtained from the analysis of several thick-walled cylinders are summarized in Tables I, II, and III. Table I is for a relatively small cylinder with Type B (Figure 1) OD notches. Fracture toughness values are given since they determine the critical crack size and thus influence the crack propagation life. The comparison of the estimated life and the actual life shows excellent agreement with many of the actual samples. The last two specimens had much longer lives than estimated. This may be the result of assuming the worst case. The actual value of the root radius was not measured for each cylinder. The minimum radius of 0.254 mm was assumed. Actual values for this dimension could be in the range from 0.254 mm to 0.508 mm. If we analyze this cylinder with a root radius of 0.508 mm, the stress concentration factor k_t is reduced from 3.0 to about 2.5. Such a reduction in k_t would increase the initiation life from 3400 cycles to 5700 cycles. Using this for initiation life increases the predicted lives of specimens 844-1R-B and 7344 to 6245 cycles and 6233 cycles, respectively, which are in much better agreement with the actual fatigue lives.

Another series of tests of the Type B configuration is presented in Table II. The agreement for these somewhat larger cylinders is not quite as good as with the small cylinders in Table I. In all cases, fatigue life is underestimated by about 5000 cycles or about 30 percent. The same argument presented above can be applied here, namely the actual value of the root radius was not measured and could vary between 0.254 mm and 0.508 mm. Using the maximum value, k_t is reduced to about 2.9 from 3.2 increasing the

TABLE I. RESULTS FOR THICK-WALLED CYLINDER WITH TYPE B EXTERNAL NOTCHES, $a = 6.56$ cm, $b_1 = 10.68$ cm, $b_2 = 11.11$ cm, Pressure = 496 MPa, Yield Strength = 1240 MPa, Root Radius = 0.254 mm, $k_t = 3.0$, 100 Percent Overstrain.

Specimen	K_{Ic} (MPa$\sqrt{m}$)	Estimated Life (Cycles)	Actual Life (Cycles)
917-3R-T	147	4151	4310
917-4R-T	152	4220	4917
7404	130	3910	3267
7414	123	3817	3948
844-1R-B	133	3945	6302
7344	132	3933	5607

TABLE II. RESULTS FOR THICK-WALLED CYLINDER WITH TYPE B EXTERNAL NOTCHES, $a = 8.26$ cm, $b_1 = 14.56$ cm, $b_2 = 15.08$ cm, Pressure = 372 MPa, Yield Strength = 1170 MPa, Root Radius = 0.254 mm, $k_t = 3.2$, 100 Percent Overstrain.

Specimen	K_{Ic} (MPa$\sqrt{m}$)	Estimated Life (Cycles)	Actual Life (Cycles)
41	180	15536	24797
51	164	15133	18006
52	139	14289	19504
54	135	14156	19993
44	165	15193	20991
45	146	14637	19862

initiation life from 12,000 cycles to 15,000 cycles. This reduces the under-estimate of actual life to about 2000 cycles or about ten percent.

The response of several Type A cylinders is summarized in Table III. This kind of notch is more severe than Type B notches as can be seen in the value of k_t. Again, the analysis method yields excellent agreement with actual behavior. The estimated life for three of the four conditions considered is within about 10 percent of the actual behavior. But the technique drastically underestimates the least severe case (smallest k_t and least percent overstrain). The reason for the discrepancy is not clear, suggesting that the method of analysis may be limited to only that class of cylinders that would fail at relatively short life.

TABLE III. RESULTS FOR THICK-WALLED CYLINDER WITH TYPE A EXTERNAL NOTCH, a = 7.75 cm, b = 13.5 cm, Root Radius = 0.762 mm, K_{Ic} = 127 MPa$\sqrt{m}$, Yield Strength = 1170 MPa, Pressure = 386 MPa.

Specimen	Notch Depth (mm)	k_t	Percent Overstrain	Estimated Life (Cycles)	Actual Life (Cycles)
D2	10.2	6.0	100	2887	3003
D3	10.2	6.0	100	2887	2452
D4	10.2	6.0	60	4255	4079
D5	10.2	6.0	60	4255	4252
S1	5.1	3.6	100	10776	9280
S2	5.1	3.6	100	10776	11224
S3	5.1	3.6	60	15862	42985
S4	5.1	3.6	60	15862	40266

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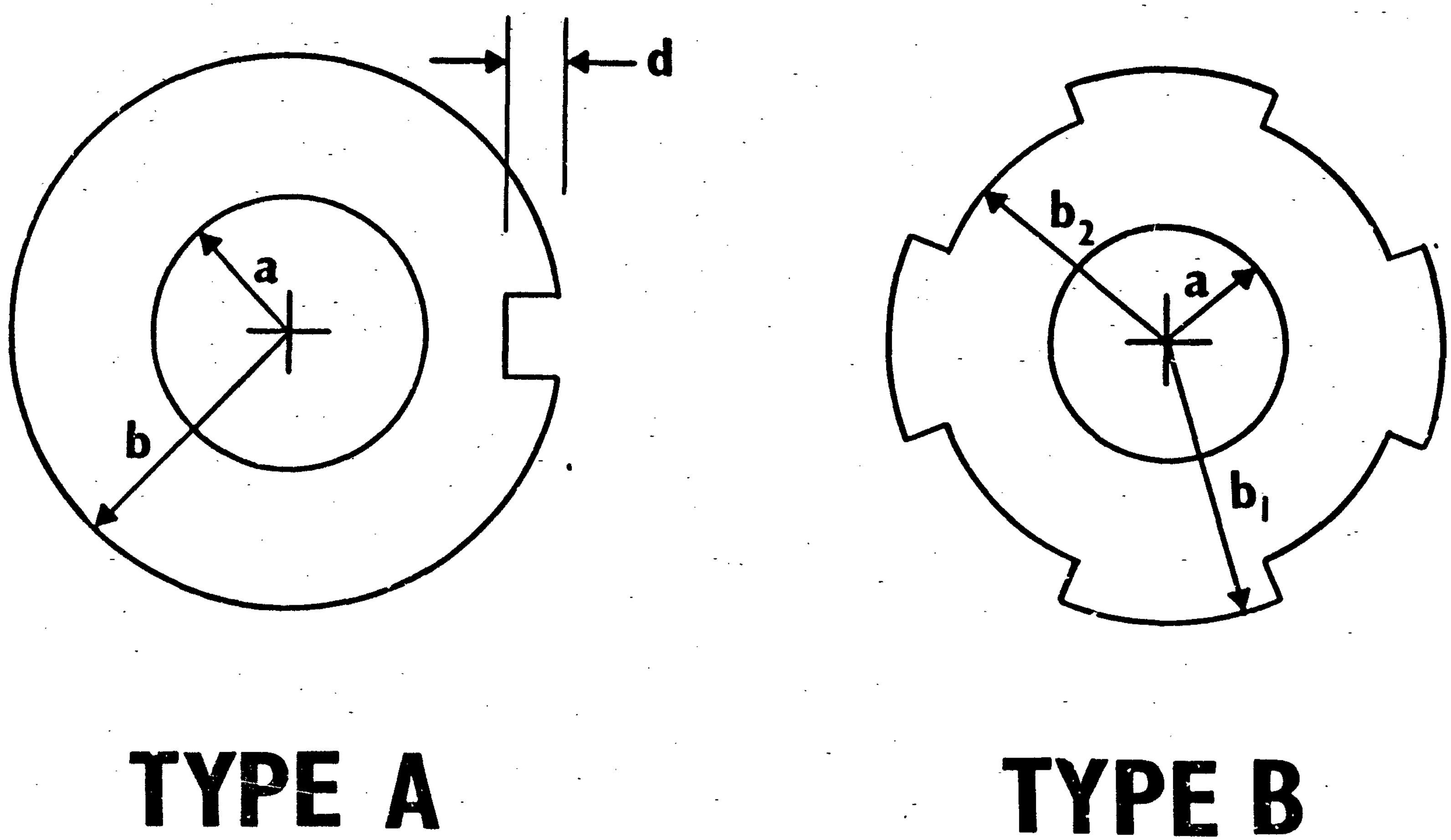

Figure 1. The two notch configurations considered, a single notch Type A) and four notches (Type B).

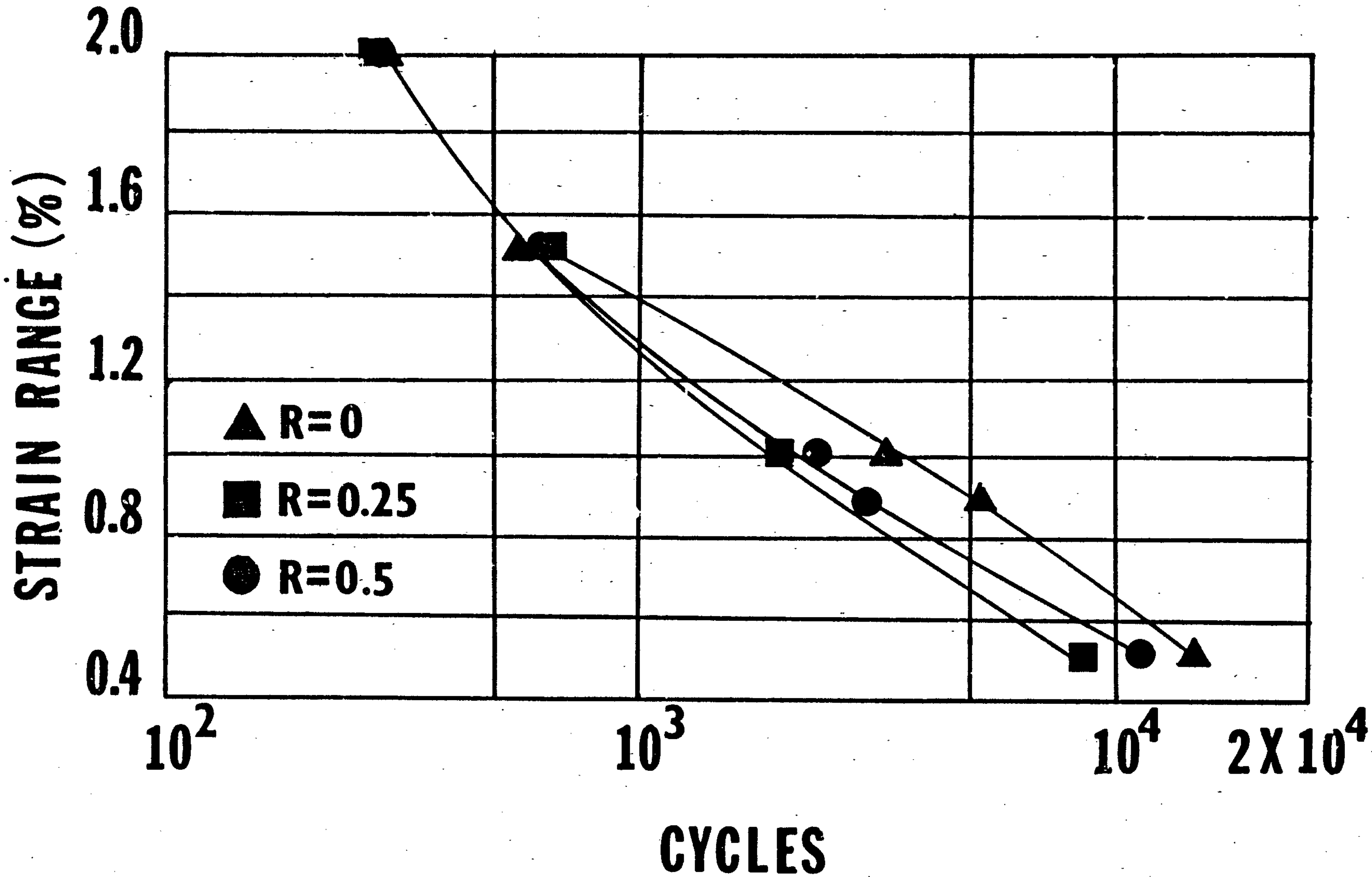

Figure 2. Low cycle fatigue data for the steel used to manufacture the cylinders studied.

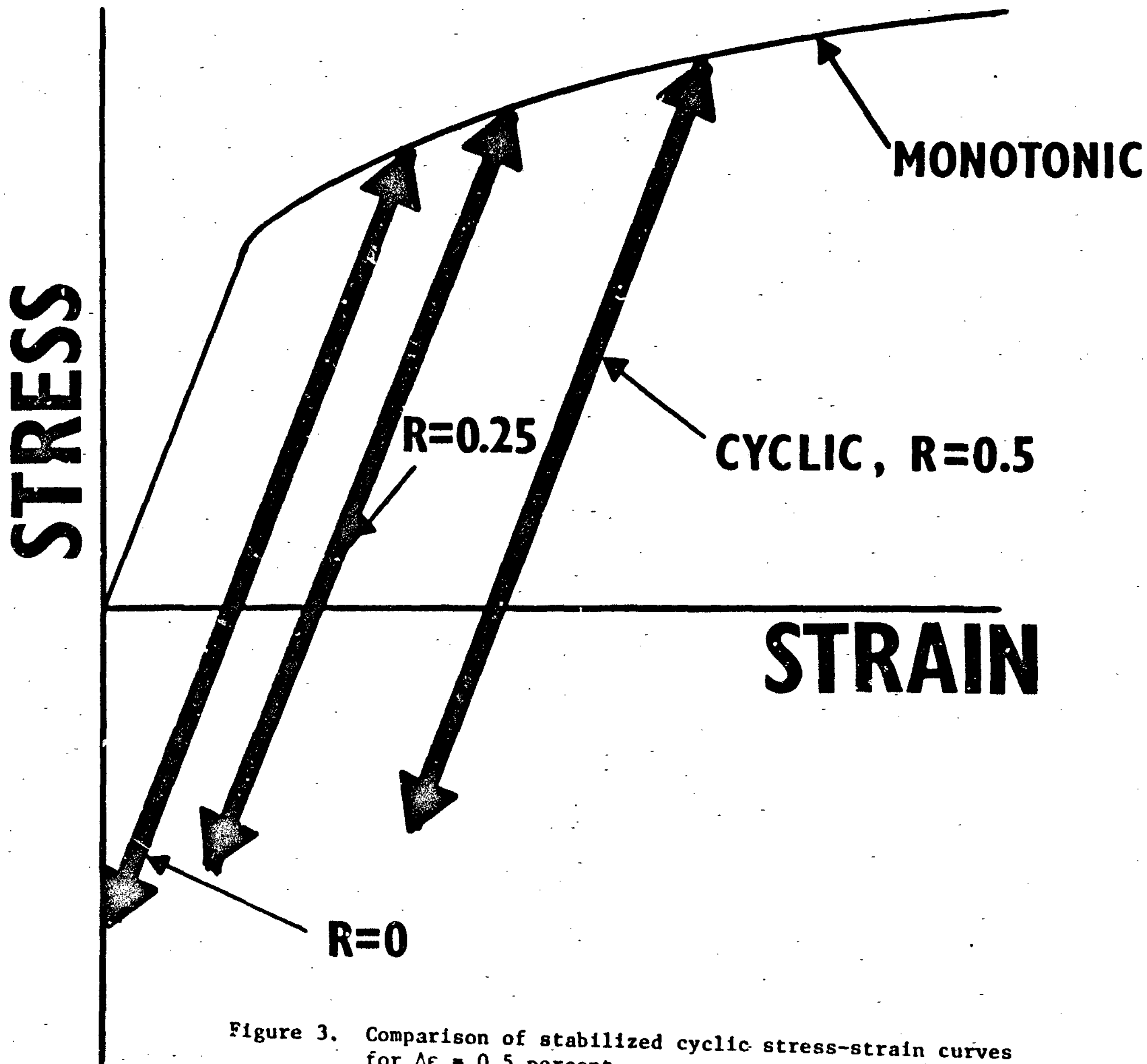

Figure 3. Comparison of stabilized cyclic stress-strain curves for $\Delta\varepsilon = 0.5$ percent.

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